



Dynamic simulation of a small modified Joule cycle reciprocating Ericsson engine for micro-cogeneration systems



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ABSTRACT

The Ericsson engine is an external combustion engine suitable for the use of certain energy sources such as solar energy, biomass and waste gases at high temperature, and thus contributes to the fight against global warming. Its open cycle configuration considered in this study can achieve good performance in low power range, most particularly in micro-cogeneration applications. The dynamic model of this engine which is the purpose of this study, takes into account both the pressure losses and the variation of the thermophysical properties of the working fluid as a function of the temperature in the system. The coded models are implemented on a Matlab/Simulink platform where the start-up dynamics and performance simulations are conducted. The optimal settings of the expansion cylinder valves, as well as the characteristic parameters of the engine are thus determined. In this configuration the engine develops a power output of 1.72 kW and reacts well when subjected to a selected perturbation.

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1. Introduction

Oil is a non-renewable source of energy whose price per barrel is continuously increasing with changes in economic and market conditions. This situation on the one hand forces motorists to adopt new spending habits and on the other hand contributes to the rise in social and political tensions worldwide. One of the challenges in the field of the thermodynamic conversion of energy consists in designing engines that are efficient, environmentally friendly and capable of using various renewable energy sources.

This is the reason why many studies are carried out in the research and scientific fields as well as in industries. In the field of dish/Stirling solar systems, the Sandia National Laboratories and Stirling Energy Systems [1] obtained an electric efficiency record of 31.25% in 2008 with a concentrated solar energy Stirling plant producing 26.75 kW in New Mexico (United-States). The gas turbine also ranks among the engines that can potentially be used to develop thermal sources of renewable energy. For CHP (combined heat and power) purposes, gas turbines generally produce

electricity ranging from 1 MW to 30 MW; micro-turbines on the other hand cover the power level up to 30 kW. A study [2] on the impact of the sizing to the performances of a micro-turbine with heat recovery suggests that the efficiency of this kind of engine is strongly deteriorated at low power range: it is estimated at 22.5% for a 5 kW installation, whereas it would have been about 27% for a typical 30 kW engine. The development of such micro-turbines (1–5 kW) being able to meet the needs of domestic CHP for example, seems to be difficult (leakage, low efficiency, cost, etc.). In this power range, the reciprocating internal combustion engine operation is expensive (especially maintenance) and noisy. The Joule cycle reciprocating Ericsson engine seems more suitable for thermal energy conversion in the low power range [3,4].

According to existing literature, some studies have reported theoretical developments on Joule cycle reciprocating engines, both on aspects of thermodynamic optimization and performance study in view of their implementation in the cogeneration sector. Indeed, Senft [5] got interested in the mechanical performances of reciprocating thermal engines by developing general mathematical models. He came out with the conclusion that among all the reciprocating engines, the ideal Stirling cycle has the maximum mechanical efficiency. Bell and Partridge [4] conducted a study

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under steady-flow conditions on an open Joule cycle reciprocating engine with regeneration, which was intended to be implemented in the cogeneration system and decentralized energy production. In a study conducted by Moss et al. [6] on a Joule cycle reciprocating engine similar to the one described above and designed for domestic micro-CHP, the problem of dimensioning was examined. The study predicts a thermal efficiency of 35% and an overall electrical efficiency of 33.2%. However, this level of performance is dependent on the relatively high values of the pressure ratio (equal to 7.5) and the engine rotational speed (1000 rpm). Wojewoda and Kazimierski [7] performed a dynamic model to describe operating conditions of an externally heated reciprocating valve engine, based on the closed Joule cycle with highly pressurized air of 25 kW, which could reach a thermal efficiency of up to 25%. However, in order to achieve this level of performance, the system requires a high rotational speed (3000 rpm) and a relatively high compression ratio (compressed air from 15 to 95 bar). It is noted that the heat transfer takes place at over 1160 K in the heater, making it difficult to achieve such an air heat exchanger for this engine. In this study, the transient problems related to the start-up and the driving of this type of engine were however not examined. More recently, Creyx et al. [8] developed a steady state model of the Ericsson engine based on modified Joule cycle which allows a thermodynamic optimization of the engine performances. Considering the modification of the thermodynamic compression and expansion cycles (isothermal and isentropic transformation) the model simulation predicts that the Joule cycle is more adapted to enhance the engine performances than the Ericsson cycle, while the adjustments of the Joule cycle (late inlet valve closing and early exhaust valve closing in the expansion cylinder) lead to the conclusion that heat transfers in the expansion cylinder wall should be enhanced for an optimized indicated mean pressure and an optimized specific indicated work, or should be avoided to improve the thermodynamic efficiency. Despite the results obtained in this study, it should however be noted that the simulated steady state model does not take into account the frictional effects, the pressure drops across the valves and in the heater.

As for all energy systems the understanding and the control of the dynamic behavior of Joule cycle reciprocating Ericsson engines are an important aspect of the development of this kind of machines [9]. In fact, it is necessary to be able to manage the transient phases (such as start-up and part load), and to guarantee the stability of operation around the nominal operating point. The objective of this work is to develop a tool for the modeling and the dynamic simulation of reciprocating Joule cycle engines which makes it possible to anticipate the possible problems of operation stability that could occur, in order to set up devices and strategies of control for such systems.

The engine configuration under study will be presented and its dynamic model developed subsequently. Before concluding this work, all the results obtained from simulations are presented and analyzed.

2. The Ericsson engine

2.1. Principle and reference cycle

A classification of heat engines has been proposed in a former study [10]. It made it possible to identify a family of specific machines: reciprocating engines with external heat supply, with separate compression and expansion cylinders, with or without regenerator, with gaseous working fluid. This family of engines includes two sub-families; the Stirling engines, working without any kind of valves (Fig. 1) and the Ericsson engines (Fig. 2) using valves at the inlet and outlet of the cylinders.

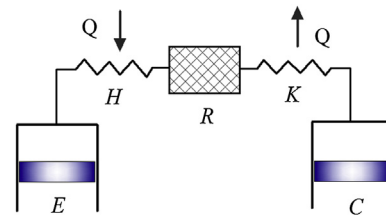


Fig. 1. Typical Stirling engine.

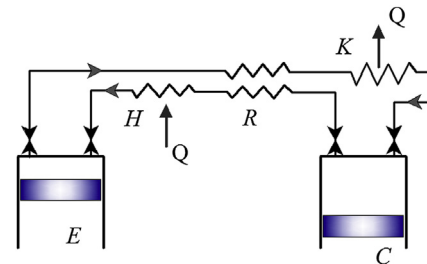


Fig. 2. Typical Ericsson engine.

From the thermodynamic point of view, the Ericsson engine is similar to a gas turbine engine where the turbo-compressor has been replaced by a reciprocating compressor and the turbine by a piston/cylinder machine. The theoretical cycle of Ericsson (2 isotherms and 2 isobars) is not adapted to describe an ideal Ericsson engine [11]. In an ideal Ericsson engine, heat transfers should take place at constant pressure while compression and expansion are supposed to be isentropic, corresponding to the Joule cycle, often used to describe the gas turbine engine principle.

2.2. Description of the studied Ericsson engine

The Ericsson engine can be operated either by a closed cycle (Fig. 2) with cooler K (in this case the system can run at high pressure and allows the use of working fluids such as helium or hydrogen), or by an open cycle with or without regeneration R. In this case, the working fluid is air that can be expanded down to the atmospheric pressure. For this study, an open Joule cycle without heat recovery, with a modified compression cycle is considered (Fig. 3). The open cycle was identified to correspond to present needs in the field of domestic micro-CHP, as well as for solar or wood-energy conversion of energy into electricity [3,12]. In these engines, the compressor and the expansion cylinder run at a low rotational speed, in order to limit pressure losses caused by valves as well as to reduce the mechanical losses. This solution ensures the best performance.

The atmospheric air is compressed by the compressor C. It receives heat Q from the thermal source through the heater H. The pressurized hot air produces work by expansion in the expansion cylinder E. The expander cylinder is externally insulated and

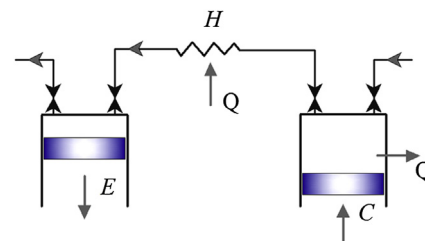


Fig. 3. Open cycle Ericsson engine without heat recovery.

operates without cooling while the compressor is cooled by the atmospheric air. The modification of Joule cycle as a result of the cooling of the compression cylinder tends to enhance the overall power output of the engine, while its efficiency tends to decrease. In fact, the decrease in the compression power output due to the cooling of the compression cylinder improves the overall power output of the engine whereas a decrease in temperature of the gas at the compressor discharge related to the same cooling process leads to a higher heat input to heater, and consequently, to a lower engine efficiency. The need to improve the power output (rather than efficiency) of the engine may be necessary when free energy sources are available, for instance: heat wastes, biomass-based agro-industrial wastes such as cocoa shells, bagasse from sugar cane, etc. A further study on the optimum pressure of the engine in this configuration would help in finding a compromise between these two opposing tendencies due to the modified compression cycle of the Joule cycle based Ericsson engine.

3. System modeling

The model of the engine is built on the basis of mass and energy balances applied to the three main components control volumes C, H and E as in the Equations (1) and (2) below. The air is used as the working fluid which is considered as an ideal gas with all its parameters supposed to be uniform in the engine components. Its thermophysical properties (specific heat, viscosity and thermal conductivity) are expressed according to the temperature (relations of Sutherland [13,14]). This model takes into account the heat transfer due to the cooling of the compressor by the surrounding air and the modeling of pressure drop due to the flow of the working fluid through the system. Moreover, gas leakages through the segmentation of both the compression and the expansion cylinders are neglected.

3.1. Compressor modeling

The mass and energy balance equations are modeled as follow:

$$\frac{dM_c}{dt} = \dot{m}_{in} - \dot{m}_{out} \quad (1)$$

$$\dot{Q}_c - \dot{W}_c = c_v(T_c) \frac{dT_c}{dt} M_c + c_v(T_c) \frac{dM_c}{dt} T_c + T_c M_c \frac{dc_v}{dt} + (c_p(T_{out}) \dot{m}_{out} T_{out} - c_p(T_{in}) \dot{m}_{in} T_{in}) \quad (2)$$

The compression work is given by the expression:

$$\dot{W}_c = p_c \frac{dV_c}{dt} \quad (3)$$

The compression process in the reciprocating compressors is accompanied by a heating of the working gas which is higher as the compression ratio increases. Resulting temperatures should be acceptable from a technological point of view, regarding the material strength, lubricating oil and usage. For these reasons, heat exchanges are taken into account between the working fluid and the internal walls of the compressor as follows:

$$\dot{Q}_c = h_c S_c (T_{wc} - T_c) \quad (4)$$

In Equation (4), the heat exchange coefficient h_c is related only to the forced convection which is predominant compared to other modes of heat transfer assumed negligible. Apart from the compressor bottom-dead-center and top-dead-center where the piston speed is zero, the gas flow remains turbulent as shown by

the simulation results presented hereafter in this paper. The heat exchange coefficient is then calculated by the expression [15]:

$$h_c = \frac{k_c}{B_c} 0.6 Re^{0.8} Pr^{0.6} \quad (5)$$

The heat exchange area is connected to the cylinder bore B_c and the piston position $x(t)$ as followed:

$$S_c = 2 \left[\frac{\pi B_c^2}{4} \right] + \pi B_c \cdot x(t) \quad (6)$$

In Equation (4), the compression cylinder wall temperature is not known. According to Kim et al. [16], the temperature of the reciprocating compressor walls does not vary by more than 0.6 K compared to its average value and is considered constant. It can then be calculated using the suction and discharge temperatures as follows:

$$T_{wc} = \frac{\int_{\tau}^{\tau+1/N} \frac{(T_{in,c} + T_{out,c})}{2} d\tau}{1/N} \quad (7)$$

Since the working fluid is considered as an ideal gas, the pressure in the compressor cylinder (cold cylinder) is determined by the ideal gas equation:

$$\frac{dp_c}{dt} = p_c \left(\frac{1}{M_c} \frac{dM_c}{dt} + \frac{1}{T_c} \frac{dT_c}{dt} - \frac{1}{V_c} \frac{dV_c}{dt} \right) \quad (8)$$

The volume of gas in that cylinder is determined according to the crank-rod kinematic as follows:

$$V_c = \frac{V_{sc}}{2} \left(1 + \lambda_c - \cos \theta - \sqrt{\lambda_c^2 - \sin^2 \theta} \right) + V_{c_c} \quad (9)$$

where $\theta = 2\pi Nt$. Usual reciprocating valves are considered for both compression and expansion. The modeling of the valves is similar to that used for internal combustion engines [17]. The air flow rate through the inlet and the exhaust valves is determined by the expression:

$$\dot{m}_j = A_j C_d j p_{high} \left(\frac{2\gamma}{(\gamma-1)r T_{high}} \left(R_j^2 - R_j^{\frac{\gamma+1}{\gamma}} \right) \right)^{\frac{1}{2}} \quad (10)$$

where $A_j = \pi L_{vj} D_{vj}$ is the so-called curtain area [17] and $R_j = p_{low,j}/p_{high,j}$. In the case where the flow velocity reaches that of the sound, the critical ratio is then evaluated as follows:

$$R_j = \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \quad (11)$$

The inlet and exhaust valve lifts are modeled by sinusoids, and determined respectively by:

$$L_{v_{in,c}} = \frac{L_{v_{max,c}}}{2} \left[1 - \cos 2\pi \left(\frac{\theta - IVO}{\pi - IVO} \right) \right] \quad (12)$$

$$L_{v_{ex,c}} = \frac{L_{v_{max,c}}}{2} \left[1 - \cos 2\pi \left(\frac{\theta - EVO - \pi}{\pi - EVO} \right) \right] \quad (13)$$

where $IVO \leq \theta \leq \pi$ and $(\pi + EVO) \leq \theta \leq 2\pi$

The closing of the inlet and exhaust valves of the compressor takes place at the bottom-dead-center and top-dead-center respectively, while their opening is delayed to angles equal to IVO (inlet valve opening) and EVO (exhaust valve opening) respectively.

3.2. Expander modeling

The expansion cylinder (hot cylinder) is considered adiabatic. The expansion piston is assembled on the same rod as the compressor piston but it moves in phase reversal compared to that of the compressor. This component is then modeled by equations similar to (1), (2), (3) and (8), (9),..., (13) with:

$$\dot{Q}_e = 0 \quad (14)$$

$$L_{v_{in,e}} = \frac{L_{v_{max,e}}}{2} \left[1 - \cos 2\pi \left(\frac{\theta}{\pi - IVC} \right) \right] \quad (15)$$

$$L_{v_{ex,e}} = \frac{L_{v_{max,e}}}{2} \left[1 - \cos 2\pi \left(\frac{\theta - \pi}{\pi - EVC} \right) \right] \quad (16)$$

where $0 \leq \theta \leq (\pi - IVC)$ and $\pi \leq \theta \leq (2\pi - EVC)$

The opening of inlet and exhaust valves of the expander takes place at the top-dead-center and bottom-dead-center respectively, while their closing are advanced by angles respectively equal to IVC (inlet valve closing) and EVC (exhaust valve closing).

These inlet and exhaust valve opening and closing angles of both cylinders are set so that the pressure equals 4 bar when the engine reaches the stable operating conditions. The other parameters are listed in the Table 1 below.

3.3. Heater modeling

In this engine as in any other hot air engine, heat is transferred from the heat source to the working fluid through a heat exchanger, whatever the origin of this heat [18]. The heater considered in this study, is a shell-and-tube heat exchanger in which the working fluid flows inside the tubes. This heat exchanger technology was chosen for its simplicity in terms of construction, durability and low cost under very severe working conditions due to high temperatures and pressure differences between the two fluids. Heat transfer between the working fluid and the hot internal walls of the heater which are supposed to have a constant temperature is determined by:

$$\dot{Q}_h = h_h S_h (T_{wh} - T_h) \quad (17)$$

In turbulent flow, the Nusselt number is calculated using the Gnielinsky correlation [19]:

$$Nu = \frac{d_t}{k_h} h_h = \frac{\frac{f}{8} (Re - 1000) Pr}{1 + 12.7 \left(\frac{f}{8} \right)^{1/2} (Pr^{2/3} - 1)} \left[1 + (d_t/L_t)^{2/3} \right] \quad (18)$$

This correlation is valid under the following conditions: $0 < d_t < L_t$; $0.6 < Pr < 2000$ and $2300 < Re < 10^6$. The friction factor f is calculated according to the Blasius relation:

$$f = 0.316 Re^{-0.25} \quad (19)$$

In case of laminar flow ($Re < 2300$), the correlation of Kays [20] is used:

$$Nu = 3.66 + \frac{0.104 (Re Pr d_t / L_t)}{1 + 0.016 (Re Pr d_t / L_t)^{0.8}} \text{ with } f = 64/Re \quad (20)$$

The pressure in the heater is calculated by integrating the ideal gas equation, while the temperature and the mass are calculated from the energy and mass balance equations. Pressure losses in the heat exchanger are calculated as follows [19]:

$$\Delta p_h = \left(f \frac{L_t}{d_t} + \xi \right) \rho \frac{u^2}{2} \quad (21)$$

3.4. Compressor and engine efficiencies

To evaluate the performance of the compression cylinder, the isothermal efficiency was preferred since the use of isentropic compression as ideal compression reference may lead, in the case of strong cooling to a ratio greater than the unity [21]. The indicated efficiency of the compressor is then calculated based on the reversible isothermal compression power and the compression power as follows:

$$\eta_{ciT} = \frac{\dot{W}_{ciT}}{\dot{W}_c} \quad (22)$$

where: $\dot{W}_{ciT} = \dot{m} \cdot (T \Delta S - \Delta h)|_{c, T=\text{const}} = -\dot{m} \cdot r T_{in,c} \ln(p_h/p_{in,c})$

The thermal efficiency of the engine is then calculated by:

$$\eta_{th} = 100 \times \frac{\dot{W}_e + \dot{W}_c}{\dot{Q}_h} \quad (23)$$

The variables-per-cycle that are used below in the simulation model results are calculated using the following formula:

$$\bar{X} = \frac{\int_{\tau}^{\tau+T} X d\tau}{T} \quad (24)$$

4. Simulation results

The Matlab/Simulink environment was used for the simulation of the models. Solver ODE23 based on an explicit Runge-Kutta (2, 3) was used with a maximum step size of 2×10^{-3} s. Therefore, it was necessary to firstly conduct several simulations of the developed models to determine the best settings of the valves. This adjustment must permit the stabilized functioning of the engine to the pressure of 4 bar with the settings indicated in Table 1.

4.1. Dynamic of starting and stability of the system

At the beginning of the simulation, pressure in the two cylinders and the heater is equal to atmospheric pressure and the

Table 1
Engine specifications.

Component	Parameters
Compressor (C)	Swept volume: $22.6 \times 10^{-4} \text{ m}^3$ Clearance volume: 11% swept volume Bore: 0.12m Connecting rod length: 0.10m Ratio of connecting rod to crank radius: 3 Opening of the inlet valve delay: 44° Opening of the exhaust valve delay: 121°
Heater (H)	Volume: 0.18 m^3 Temperature of wall: 873°K Heat exchange area: 8 m^2
Expander (E)	Swept volume: $42.5 \times 10^{-4} \text{ m}^3$ Clearance volume: 8% swept volume Bore: 0.16m Connecting rod length: 0.104m Rod to crank radius ratio: 3 Closing of the inlet valve advance: 113° Closing of the exhaust valve advance: 30°
Rotational speed: 480rpm	

temperature is equal to 300 K. The start-up of the engine begins with the assumption that the temperature of the internal walls of the heater's tubes is constant and equals the fixed value of 873 K. This then implies a hot start when wall temperature of the main components is considered as a constant. Fig. 4 presents the three-phase dynamic of the engine's start-up, illustrated by the aid of the evolution as a function of time of the difference between the hot gas flow admitted into the expansion cylinder $\dot{m}_{in,e}$ and that discharged by the compressor into the heater $\dot{m}_{out,c}$. The first phase (part of the curve from I to II in Fig. 4) corresponds to the pressurization of the engine (at the expense of some mechanical external energy source). During this phase that lasts only 0.36 s, the expansion cylinder is isolated from the heater, the inlet valve being maintained closed and the exhaust valve being maintained open. Fig. 4 shows that the part of the curve corresponding to the pressurization of the engine is negative, and presents the oscillations that translate the increase in pulsating flow rate of the gas discharged from the compressor to the heater. When the pressure reaches the adopted level for starting the expander (set at 3 bar), the second start-up phase (part of the curve from II to III in Fig. 4) begins with the release of the expander valves. The corresponding part of the curve grows with a positive slope. This means that the flow rate of the working fluid admitted into the expansion cylinder increases progressively and produces work. Following that, there is equality between the flow rate discharged from the compressor and that admitted into the expander. This equilibrium corresponds to the engine's third operating phase (part of the curve from III to IV in Fig. 4) where the functioning parameters are stabilized, after 39 rotation cycles.

Figs. 5 and 6 are illustrations of the evolution of temperature and pressure in each of the compartments of the engine after an appropriate setting of the valves has been done to enable the complete re-compression of residual gas in the hot cylinder in order to meet the steady flow operation condition of the system at a pressure of 4 bar.

In this permanent state, the engine generates an output of 1716 W and a thermal efficiency of about 23%. The simulations enable us to predict the impact of temperature dependence of certain thermophysical properties of air in the system, as it happens to the thermal conductivity which increases practically by 68% in the heater, due to the level of temperature existing there. This contributes practically to the doubling of the intensity of convective heat exchange along the walls of this heat exchanger.

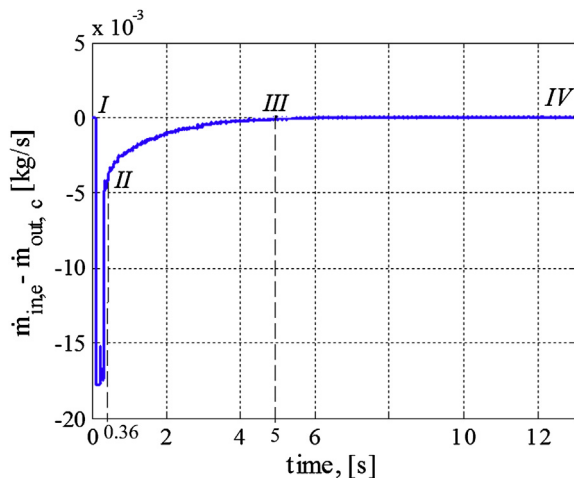


Fig. 4. Dynamic of the mass flow rate difference between the expander and the compressor.

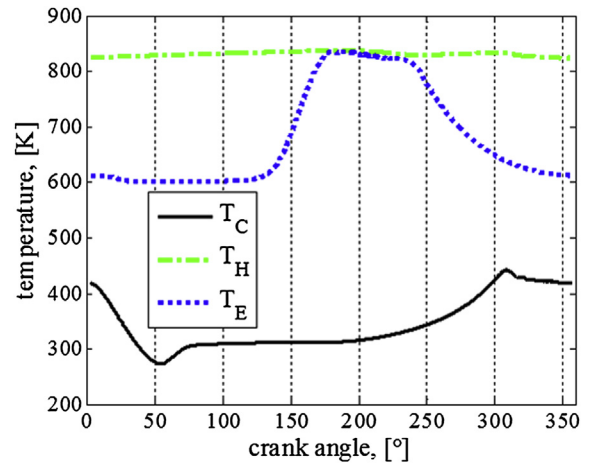


Fig. 5. Temperature as function of crank angle.

Fig. 7 illustrates the compressor's pV diagram where pressure losses are simulated with the flow of gas through the two valves. These pressure losses, which depend on the geometry and the dynamics of the opening of the valves, have an impact on the system's performance.

In fact, the overpressure at discharge valve illustrated by Δp_{out} contributes to a supplementary work and an elevation of temperature of the discharged gas, while the pressure drop at the inlet valve materialized by Δp_{in} causes the reduction of the flow rate, an additional work and an increase of the temperature at the discharge [22].

Taking into account the fields of application considered in this work, the rotational speeds envisaged for the studied engine are low, less than 1000 rpm. When the engine turns at the adopted speed (that correspond to an average piston velocity of 3.2 m/s), the inlet valve causes a pressure drop of 20 kPa, while the pressure at the discharge also drops by 32.5 kPa. Apart from the top-dead-center and the bottom-dead-center where the speed is zero, the flow of gas is turbulent with a heat exchange coefficient whose average value is equal to 310 W/m² K per cycle.

The indicated efficiency of the compressor has been calculated based on Equation (22) when the engine operates in steady state for different increasing values of the rotational speed in order to investigate its behavior in this speed range. Fig. 8 shows the

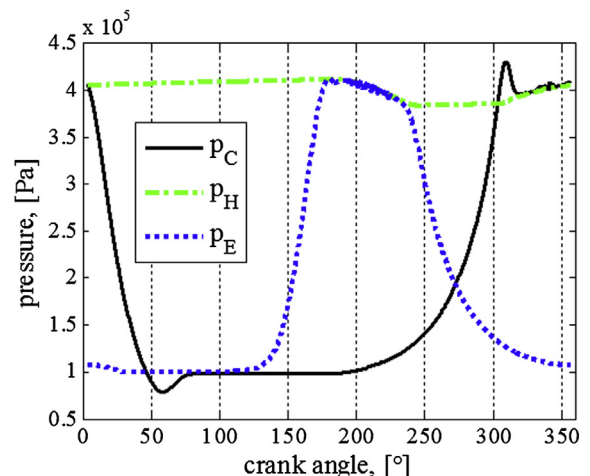


Fig. 6. Pressure as function of crank angle.

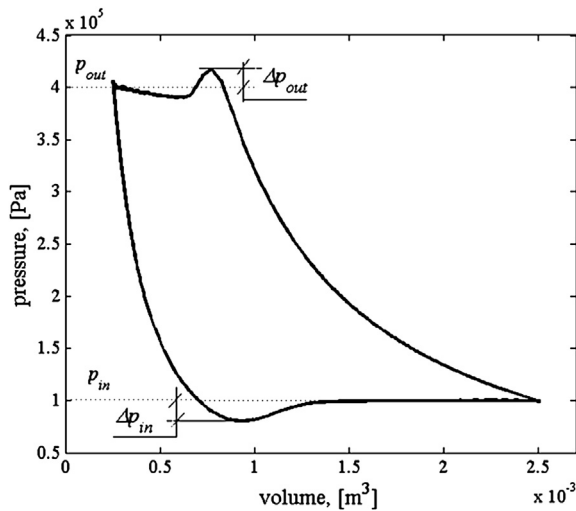


Fig. 7. pV plot including pressure drop in the compressor valves.

evolution of the compressor's isothermal efficiency, which drops with the increasing values of the rotational speed. In fact, the reversible isothermal compression power, as well as the calculated compression power, grows with increasing values of the rotational speed, according to Equation (22). However, the compression power takes into account that losses from both the pressure drops across the valves and from the inefficient cooling system grow faster than the reversible isothermal compression power. It follows, therefore, that the efficiency of the compression process, which is nothing more than the ratio between the two quantities, will decrease when the rotational speed of the engine increases.

However, in the heater the low rotational speed adopted in this study, as justified above, causes a very low air flow rate inside the heater tubes (2.8 m/s) such that both under the transient and steady conditions, the air flow is laminar. The intensity of the heat exchanges characterized by the convection coefficient (which on average is 21.7 W/m² K) only varies depending on the temperature inside the heater. This can be explained by the fact that the air thermal conductivity rises with the increase of the temperature in the heater. Pressure losses in the heater (about 196 Pa in each of the heater tubes under steady flow conditions) are insignificant compared for instance, to the pressure drops at the compressor valves.

Since the expansion work is produced using the adiabatic process in the hot cylinder, the overall output of the engine, which is the difference between the expander power output and the compression power input is increased due to the cooling of the compressor. This cooling system contributes to reduce the power that is necessary to compress and discharge the working gas into the heater. However, the cooled compression process (with resulting lower discharge temperature) causes an increase in heat transfers into the heater. This situation is coupled with the fact that this trend is exacerbated by the variation of air conductivity as a function of the temperature as indicated above. It is thus very obvious that, besides the power gain resulting from the cooled compression process, the performance of the system as a whole drops.

In order to increase the system efficiency, the temperature of the gas downstream the compressor should be raised, for instance by enhancing the hot gas at the outlet of the hot cylinder where the temperature of about 628 K is higher than that of compressor discharge gas, equal to 344 K. It is, indeed well-known that, for the low compression ratios (as the case considered in this work) only the open Joule cycle with heat recovery makes it possible to obtain good thermal efficiencies.

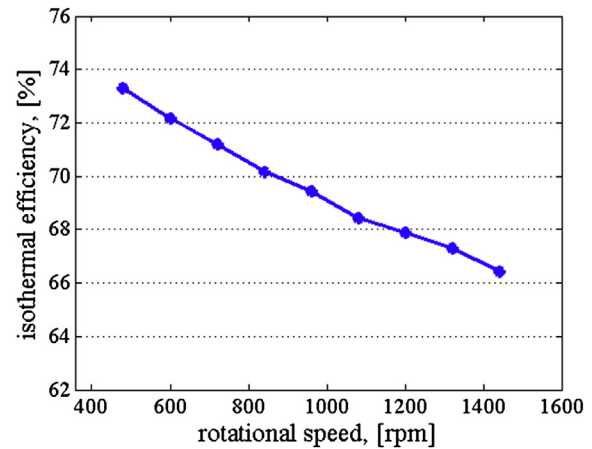


Fig. 8. Isothermal efficiency of the compressor as function of rotational speed.

4.2. Perturbation influences on the system

There are many kinds of perturbations that can be modeled and simulated, in order to analyze their influences on the stability of the system as well as their impacts on the performances of the engine under stable operating conditions. The engine can then go through various transient phases, depending on the perturbations to which it is subjected. The perturbations could be for instance, a compressor inlet pressure drop, a hot source temperature drop, a delay of closing of the expansion cylinder inlet valve, etc.

Once the engine has started and stable operation are established, a sudden drop of 10 kPa is induced to the inlet pressure of the compressor and simulated in order to study the transient dynamics associated with this perturbation, its influence on the stability of the system as well as its impact on the performances of the engine. Such a drop is practically possible and could for instance be linked to air intake filter plugging. This phenomenon is simulated on 120 engine rotation cycles. The simulation results are shown in Figs. 9–11. The two curves in Fig. 9 show that at the end of the perturbation the compressor indicator diagram moves downwards, after having cross a transient phase as depicted in the other figures. The resulting discharge pressure decreases as consequence. It happens that the inlet valve, whose setting was not changed, opens too late, causing more pressure drop during the inlet phase. The discharge valve, whose setting was also not modified, opens too early; thereby significantly reduces the discharge pressure.

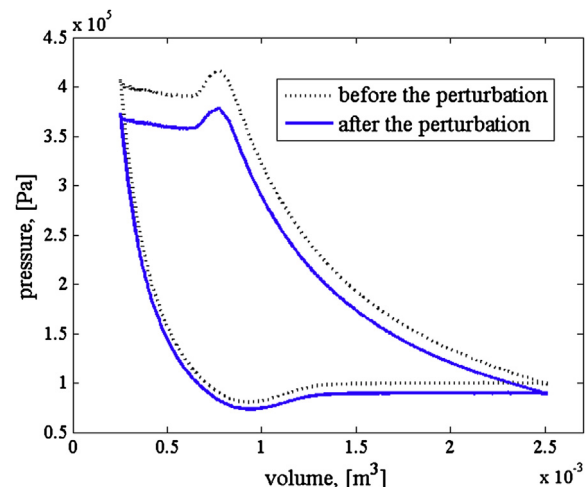


Fig. 9. Compressor pV plot under stable operating conditions of the engine.

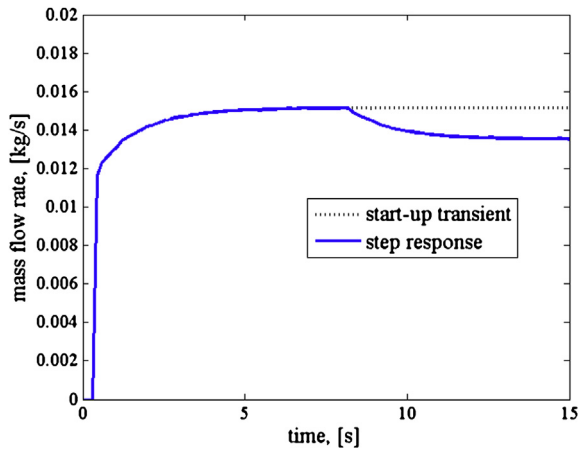


Fig. 10. Dynamics behavior of the engine mass flow rate.

It should be noted that during the time when the compressed air is discharged outside the compressor, the pressure in the compression cylinder is not constant. The compressor discharges into the heater whose volume remains constant, its pressure during the discharge period increases until the closing of the outlet valve, whereas the pressure in the expansion cylinder drops during the inlet phase of this component (Fig. 6). These two tendencies whose evolution is prejudicial with the performances of the engine are due to the fact that the volume of the heater is not as large as necessary if compared with that of the expander.

In order to compensate this phenomenon without modifying the initial configuration of the pistons of the two cylinders, it is enough either to increase the volume of the heater so that the V_h/V_e ratio rises sufficiently, or to envisage a buffer volume upstream the heater. This solution enables the rectification of the incriminated sections of the curves but the engine thus modified becomes less compact and the start-up transient duration necessarily increases.

5. Conclusion

A dynamic model of an Ericsson engine based on the open Joule cycle with air cooled compression cylinder was coded and simulated. The simulation results predict a rapid hot start dynamic (achieved in 5 s) and a stable operation of the engine with thermal efficiency of about 23%. In this low-speed two stroke engine, the pressure losses in the heater, where the air flow remains laminar

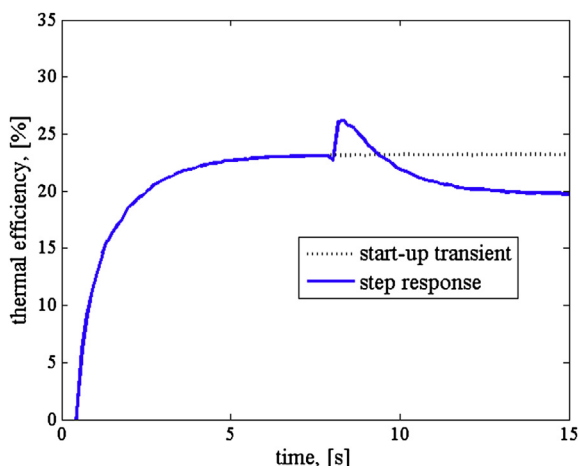


Fig. 11. Dynamics behavior of the engine thermal efficiency.

are insignificant compared to the pressure drop at the compressor valves. Subjected to a selected perturbation (compressor inlet pressure drop) the engine operation is stabilized after a rapid transient phase but undergoes, logically a drop of efficiency.

These results, as to what concerns stability and performances, show that this technologically simple engine should be suited to small CHP systems. Furthermore, the use of a regeneration cycle in view of recovering the waste heat of hot gases downstream the expander would permit to improve the engine performances. A better appreciation of the dynamics of operation will however be made by taking into account the heat transfer and the thermal inertia in the heater.

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Nomenclature

A: flow area of the valve, [m²]
 B: cylinder bore, [m]
 C: compression cylinder
 Cd: discharge coefficient
 CHP: combined heat and power
 cp: specific heat of air at constant pressure, [J/kg·K]
 cv: specific heat of air at constant volume, [J/kg·K]

d: diameter, [m]
D_v: valve head diameter, [m]
E: expansion cylinder
EVC: exhaust valve closing, [rad]
EVO: exhaust valve opening, [rad]
f: friction coefficient
h: heat exchange coefficient, [W/(m²°K)]
H: heater
IVC: inlet valve closing, [rad]
IVO: inlet valve opening, [rad]
k: thermal conductivity, [W/(m°K)]
K: cooler
L: length, [m]
L_v: valve lift, [m]
M: mass, [kg]
m: mass flow rate, [kg/s]
N: rotational speed, [rev/s]
p: pressure, [Pa]
Q: thermal power, [W]
Q: heat, [J]
r: gas specific constant, [J/kg°K]
R: recuperator
rpm: rotation per minute
S: heat exchange surface area, [m²]
t: time, [s]
T: temperature, [K] or period, [s]
u: speed, [m/s]
V: volume, [m³]
V_c: clearance volume, [m³]
V_s: swept volume, [m³]
W: power, [W]
x: piston position in the cylinder, [m]

$\bar{X}$: mean value for the parameter *X*

Greeks symbols

γ : specific heat ratio
 η : thermal efficiency, [%]
 λ : ratio of connecting rod length to crank radius
 ν : kinematic viscosity, [m²/s]
 θ : crank angle, [rad]
 ρ : density, [kg/m³]
 ξ : loss coefficient
 Δp : pressure loss, [Pa]

Subscripts

c: compressor
e: expander
ex: exhaust
h: heater
high: high-side
in: into the control volume, inlet
is: isentropic
iT: isothermal
j: component (compressor or expander)
low: low-side
m: mean value
out: out of the control volume
p: piston
t: tube
th: theoretical
w: wall